

Theoretical simulation and experimental research on the system of air source energy independence driven by internal-combustion engine

Chen Yiguang, Yang Zhao*, Wu Xi, Wang Mingtao, Liu Huanwei

School of Mechanical Engineering, Tianjin University, 92 Weijin Road, Tianjin 300072, PR China

ARTICLE INFO

Article history:

Received 27 April 2010

Received in revised form 20 January 2011

Accepted 24 January 2011

Keywords:

Energy independence driven by
internal-combustion engine (EIICE)

Mathematical model

Performance

Experimental research

ABSTRACT

Presents a new system of air source energy independence driven by internal-combustion engine (EIICE), which used natural gas or other fuels as an independent input energy, and could provide the heating, cooling and hot water for the buildings efficiently. It also could provide electricity for electric equipments of the system. The performance of air source EIICE system was investigated theoretically and experimentally. The experimental and simulation results indicated that the heat capacity of plate heat exchanger (P-HE), heat recovered from exhaust gas heat exchanger (EG-HE), input power of compressor, output power of engine and fuel consumption increased with the increase of the rotary speed, water flow rate of the P-HE and evaporation temperature. Heat recovered from the cylinder jacket heat exchanger (CJ-HE) increased with the increase of the rotary speed and evaporation temperature, but decreased with the increase of the water flow rate of P-HE. The coefficient of performance (COP_t) and primary energy ratio (PER_t) of air source EIICE system also increased with the increase of the water flow rate of P-HE and evaporation temperature, but decreased with the increase of the rotary speed.

© 2011 Elsevier B.V. All rights reserved.

1. Introduction

China is one of the countries that take coal as the main energy resource and most of the coal is used directly for burning without coal-gasification, which caused lots of environmental problems. Monitoring result shows that most of the soot and SO_2 in the atmosphere are produced by burning coal. Therefore, in recent years, many countries pay more attention to the clean energy, such as the natural gas, which can not only decrease the usage of coal and oil but also reduce the environmental pollution. Many investigators are focused on the utilization of this abundant energy. The gas engine-driven heat pump (GEHP) is a kind of new energy-efficient heating and air conditioning equipment which consumes natural gas. The GEHP can make full use of the waste heat from the internal-combustion engine and achieve a higher PER than other forms of heating/cooling systems, therefore it has been considered by an increasing number of people as a preferable choice in the heating and air-conditioning scheme. Since the GEHP can take advantage of this high-quality clean energy efficiently, extending its use would be very important to the energy-saving and environment protection. Rational design and utilization of the internal-combustion engine heat pump has become a common subject around the world.

Many researches have been done on the technologies of the internal-combustion engine heat pump. Some investigators had

devoted their attention to the system integral energy efficiency and economic aspects of the units. For example, Vinton and Getman [1] made their tests in a climate of southern America and measured the overall COP as 1.37 for heating and 1.15 for cooling. Xu and Yang [2] established the mathematical model for the second heat exchanger and investigated the influence of various factors including the outdoor air temperature and humidity in summer and winter of northern China. Sanaye et al. [3] made an economic analysis of GEHP for residential and commercial buildings in various climate regions of Iran. Another important research area was the system modeling of the GEHP. Zhang et al. [4] established a steady-state model, which contained an experimental gas engine waste heat model, and also a theoretical heat recovery model. Yang and Zhang [5] improved a dynamic model. This model included an EG-HE. The other investigators had devoted their attention to improving the control strategies of the GEHP systems. For example, Li et al. [6] improved a cascade fuzzy control strategy, which was a stable control system with reduced reaction time and a little overshoot in temperature. Yang et al. [7] presented an intelligent control simulation to research the dynamic characteristics of the GEHP system. The results showed that the model was very effective in analyzing the effects of the control system, and the steady state accuracy of the intelligent control scheme was higher than that of the fuzzy controller. In another aspect, investigations on improving the performance of the GEHP systems had also been made. For example, Li et al. [8] designed a hybrid-power GEHP system. Their simulating results of the power system showed that for a conventional GEHP system, the maximum and minimum thermal

* Corresponding author. Tel.: +86 22 2789 0627; fax: +86 22 2789 0627.

E-mail address: zhaoyang@tju.edu.cn (Z. Yang).

Nomenclature

A	area (m^2)
A_p	the primary surface area between the finned tube (m^2)
A_f	the finned surface area (m^2)
a	void fraction
B	natural gas consumption (kg/s)
C_d	flow coefficient
C_p	specific heat at constant pressure ($\text{kJ}/(\text{kg K})$)
C_{p_w}	specific heat of water ($\text{kJ}/(\text{kg K})$)
C_v	specific heat at constant volume ($\text{kJ}/(\text{kg K})$)
d_{eq}	equivalent diameter (m)
h	specific enthalpy (kJ/kg)
I	electric current (A)
k	polytropic exponent
M	mass flow rate (kg/s)
m	quality (kg)
N	compressor speed (rpm)
n	compression index
P	pressure (Pa)
Q	heat transfer rate (kW)
Q_B	total heat from natural gas (kW)
Q_c	the heat of P-HE (kW)
Q_{cj}	the heat recovered from the cooling water of the engine (kW)
Q_e	effective shaft work (kW)
Q_{fuel}	the heat produced by burning fuels (kJ)
Q_g	the exhaust gas heat (kW)
Q_s	heat loss (kW)
R	gas constant ($\text{kJ}/(\text{kg K})$), resistance (Ω)
S_w	tube wall perimeter (m)
T	temperature (K)
t	temperature ($^{\circ}\text{C}$)
u	specific internal energy (kJ/kg)
U	voltage (V)
V	volume (m^3)
W	power (W)
Z	the length of pipe (m)

Greek symbols

α	heat transfer coefficient ($\text{W}/(\text{m}^2 \text{K})$)
φ	crank shaft angle ($^{\circ}\text{CA}$)
ρ	density (kg/m^3)
ξ	deposition coefficient
η	efficiency
$\eta_{f\bar{\xi}}$	fin efficiency of wet condition
τ_w	wall shear stresses (Pa)
λ	thermal conductivity ($\text{W}/\text{m}^{\circ}\text{C}$)

Subscripts

a	air
c	condenser
d	d -axis
cp	compressor
cj	cylinder jacket
cy	cylinder
e	evaporator
ev	expansion valve
f	fin
g	gas
gen	generator
in	inlet
l	liquid

out	outlet
p	wall
pum	pump
q	q -axis
r	refrigerant
v	vapor
w	water
0	0 -axis

efficiencies of the power system were 33% and 22%, respectively, while those of the novel hybrid-power GEHP system were 37% and 27%. Yagyu et al. [9,10] designed a gas engine driven stirling heat pump in Japan and tested its performance. The results showed that if the heat pump system could be pressurized up to 5 MPa, COP would be improved from 1.9 to 2.42.

Despite of a large amount of investigators had devoted their attention to the above mentioned research areas of the internal-combustion engine heat pump system, the published literatures for study on the performance of air source EIICE are very limited. The system of air source EIICE is a kind of energy-saving and environment-friendly equipment, which uses natural gas or other fuels as an independent input energy, and could provide heating, cooling and hot water for buildings. It also can provide electricity for electric equipment of the system itself. Compared with the common heat pump, the air source EIICE holds the following advantages: first of all, it can be run without the support of power net and avoiding the effect of power outage. Secondly, waste heat recovered from the engine cooling water and exhaust gas can be used efficiently, thus it has a higher PER than the general electrical heat pumps. What is more, it also contains the merits of stepless variable speed, higher seasonal energy efficiency ratio (SEER) and narrowing the gaps between the requirement and provision of electricity and natural gas in summer and winter. Therefore, the purpose of this study is to theoretically and experimentally investigate the performance of air source EIICE. Based on these simulation and experimental data, the effects of variable rotary speed, variable water flow rate of P-HE and variable evaporation temperature on the performance of air source EIICE have been discussed.

2. The simulation model of air source EIICE

Three mathematical system models for the internal-combustion engine, heat pump and autonomous power supply had been developed in this study.

2.1. Internal-combustion engine system model

2.1.1. Internal-combustion engine model

Regarding the cylinder as a thermodynamic system, the boundary of system consists of the cylinder episorium, cylinder head and piston top surface, and the thermal system meets the energy and mass balance equation, in addition, the state equation of working substance in the cylinder must be supplied. The gaseous working substance in the cylinder can be regarded as an ideal gas. Therefore, the ideal gas equation of state can be added. The internal-combustion engine is the source power of the whole system, and the compressor and generator are driven via the belt. The engine and the compressor/generator can be well coupled by the parameters of speed (rpm) and power (W). Parameter φ means the rotation angle ($^{\circ}\text{CA}$) of the crank shaft. In the calculating model, powers (W) would be equal by adjusting the throttle percentage of the engine at a constant speed. In order to simplify the analysis, the following assumptions have been made:

- (1) Regarding the working substance as ideal gas, and uniform distribution.
- (2) When the gas inflow or outflow the cylinder, it can be regarded as the steady-state flow.
- (3) The kinetic energy of the inflow and outflow is negligible.

The mass conservation equation is:

$$\frac{dm}{d\varphi} = \frac{dm_{cy,in}}{d\varphi} + \frac{dm_{cy,out}}{d\varphi} \quad (1)$$

The energy conservation equation is:

$$\frac{dT}{d\varphi} = \frac{1}{m \cdot c_v} \left[\frac{dQ_B}{d\varphi} + \frac{dQ_{cy}}{d\varphi} - P \frac{dV_{cy}}{d\varphi} + (h_{cy,in} - u) \frac{dm_{cy,in}}{d\varphi} + (h_{cy,out} - u) \frac{dm_{cy,out}}{d\varphi} \right] \quad (2)$$

The ideal gas equation of state is:

$$PV = mRT \quad (3)$$

The above mentioned equations constitute the general equation group of solving working substance state in the cylinder. The changes of the refrigerant temperature, pressure and quality in the circulation process can be obtained by solving these equations. And then the actual working process of the internal-combustion engine can be simulated.

2.1.2. Engine waste heat recovery system model

The model of heat exchangers for recovering the waste heat can be described by the energy equilibrium [11] of the internal-combustion engine:

$$Q_B = Q_e + Q_{cj} + Q_g + Q_s \quad (4)$$

The heat recovered from the cooling water of the engine is:

$$Q_{cj} = M_{cj} C_{pw} (T_{cj,out} - T_{cj,in}) \quad (5)$$

The exhaust gas heat is given by:

$$Q_g = (B + M_a)(C_{pg} T_{g,out} - C_{pa} T_{g,in}) \quad (6)$$

2.2. Heat pump system model

2.2.1. Compressor model

The mass of the refrigerant at the inlet and outlet of compressor is the same, and regarding the working process of compressor as an adiabatic process, therefore, the mass flow rate M_{cp} can be calculated by the following equation:

$$M_{cp} = N \cdot V_{cp} \cdot \rho_{cp,in} \left[0.98 - 0.085 \cdot \left[\left(\frac{P_c}{P_e} \right)^{1/n} - 1 \right] \right] \quad (7)$$

The outlet enthalpy of compressor can be calculated by:

$$h_{cp,out} = h_{cp,in} + P_e \cdot \frac{k}{k-1} \cdot \left[\left(\frac{P_c}{P_e} \right)^{k-1/k} - 1 \right] \cdot (\rho_{cp,in} \cdot \eta_{cp})^{-1} \quad (8)$$

2.2.2. Heat exchangers model

In this air source EIICE system, a plate exchanger is used for indoor heat exchanger, and a finned tube exchanger is used for outdoor heat exchanger. The heat exchanger is a major part of the entire EIICE system, and the transfer processes of the heat and mass are very complicated. In order to improve the speed and stability of calculation, the models of plate exchanger and finned tube exchanger are established by the method of distributed parameter in this paper. The entire heat exchanger is divided into a number of control volumes, and the corresponding control equations are

established by the lumped parameter method in each volume to solve the changes of its internal heat transfer and pressure.

The governing equations for each control volume can be expressed as follows:

The mass conservation equation is:

$$\frac{\partial}{\partial z} [a \rho_v u_v + (1-a) \rho_l u_l] = 0 \quad (9)$$

The energy conservation equation is:

$$\frac{\partial}{\partial z} [a \rho_v u_v h_v + (1-a) \rho_l u_l h_l] = q_r \quad (10)$$

The momentum conservation equation is:

$$\frac{\partial}{\partial z} [a \rho_v u_v^2 + (1-a) \rho_l u_l^2] = -\frac{\partial P}{\partial z} - \frac{\tau_w S_w}{A} \quad (11)$$

The following correlations are used for heat transfer computation of the plate heat exchanger: for the condenser, the correlation of Yan [12] is used to calculate the condensing heat transfer coefficients in the single phase flow region and two phase flow region. While for the evaporator, the evaporating heat transfer coefficients of the single phase region inside a horizontal smooth tube and two phase flow region in the plate heat exchanger are represented by Muley [13] and Yan [14], respectively.

The following correlations are used for heat transfer computation of the finned tube heat exchanger: for the evaporator, the heat transfer coefficient of two phase region is adopted from the literatures [15,16]. While for the condenser, the correlation of Soliman, the correlation of Tandon and the correlation of Akers and Rosson [17] are adopted to calculate the condensing heat transfer coefficient of mist flow, annular flow and wave flow, respectively. In the two phase flow region, and in the single phase flow region (including evaporator) of the finned tube heat exchanger, the correlation of Dittus-Boeler [18] is used to calculate the heat transfer coefficient.

For the single phase flow region inside a horizontal smooth tube, there are three unknown variables and three equations, therefore, the three variables can be obtained by solving the three equations. For the vapor-liquid region inside a horizontal smooth tube, there are four unknown variables and three equations, in order to solve these equations, a complementary equation must be added. In this paper, the slip ratio model was selected from previous research suggested by Premoli [18].

Water side (plate type) energy conservation:

$$M_r \alpha_r (t_r - t_p) - M_w \alpha_w (t_p - t_w) = 0 \quad (12)$$

Air side (finned tube type) energy conservation:

The air side heat transfer condition can be divided into dry condition and wet condition. The air side pressure drop could be ignored, so the governing equations for air side are as follows:

The energy conservation equation under dry condition is:

$$dQ_a = M_{a,in} h_{a,in} - M_{a,out} h_{a,out} = \alpha_a (\eta_f dA_f + dA_p) (t_a - t_p) \quad (13)$$

The energy conservation equation under wet condition is:

$$dQ_a = M_{a,in} h_{a,in} - M_{a,out} h_{a,out} = \xi \alpha_a (\eta_{fg} dA_f + dA_p) (t_a - t_p) \quad (14)$$

2.2.3. Expansion valve model

It can be regarded as an isenthalpic process that the refrigerant flows through the expansion valve, and there is on refrigerant mass variation in this process. Therefore, the energy equation can be simplified as:

$$h_{ev,in} = h_{ev,out} \quad (15)$$

In this paper, the expansion valve model was used of the empirical formula [19] as follow:

$$M_{ev} = C_d A_{ev} \sqrt{2 \rho_{ev,in} (P_{ev,in} - P_{ev,out})} \quad (16)$$

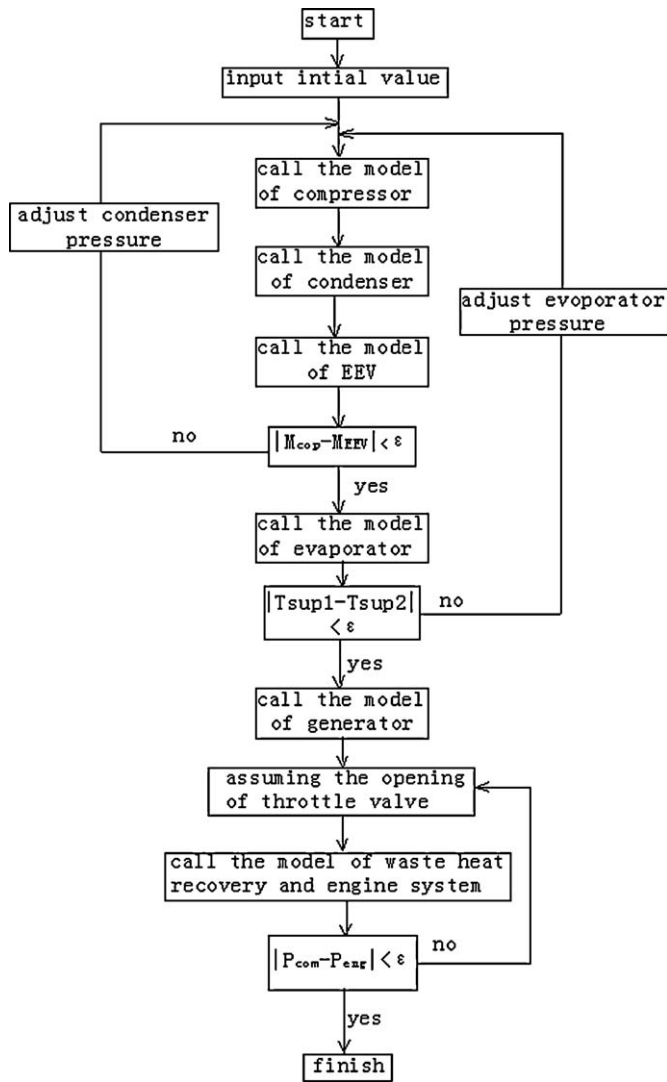


Fig. 1. The flowchart of numerical algorithm.

Table 1

The component parameters of the test EIICE unit.

Parameters	Values
Finned tube exchanger	
Tube length	2000 mm
Fin type	Wavy fin
Out tube diameter	10 mm
Fin thickness	0.2 mm
Fin pitch	2.5 mm
Tube space	35 mm
Plate exchanger	Water
Flow rate	7193 kg/h
Total heat transfer area	5.68 m ²
Step tablet	100
Flow pattern	Countercurrent
Open compressor	
Model	6NFC(Y)
Refrigerant	R134a
Model	Alco-EX5
Internal combustion engine	
Model	CA4G25E-N
Number of cylinders	4
Cylinder diameter/strokes	87.5/104 mm
Output volume	2.5 L

3. Experimental apparatus

The schematic diagram of EIICE system and measuring point are shown in Fig. 2. The EIICE system is mainly composed of a reversible vapor compression heat pump system, the cold and heat source system, the data acquisition and controlling system, the autonomous power supply system and the internal-combustion engine system. The core of the system is the compression heat pump driven by an internal combustion engine instead of an electric motor. The reversible vapor compression heat pump system includes an open compressor, a condenser, an expansion valve and two evaporators. There is also a receiver in the system to balance the difference of the refrigerant in the heating and cooling mode. A CB52-X plate heat exchanger and a finned tube heat exchanger are used as the condenser and evaporator, respectively. An open compressor is used in the EIICE system. The stepless speed regulation of the open compressor can be realized when the speed lies between 500 rpm and 3000 rpm. And the Alco-EX5 electronic expansion valves have been introduced into the system. The internal combustion engine system consists of the fuel supply system, an internal combustion engine, a heat recovery system of smoke evacuation and cylinder heat recovery system. The data acquisition and controlling system is composed of thermoelectric couple, pressure transducer, rotate speed transducer, flow rate transducer, voltage transducer, electric current transducer, PLC (Programmable Logic Controller) module, rotating controller, expansion valve controller and water outlet temperature controller. The cold and heat source system is mainly comprised the evaporating fan, condenser circulation pump, water tank, relevant pipelines and so on. The autonomous power supply system consists of synchronous generator, rectifier, voltage regulator, storage battery and so on. The autonomous power supply system is mainly offering the electricity for the electric equipment of the system. The EG-HE is a plate-fin heat exchanger. The component parameters of the tested EIICE unit are presented in Table 1.

4. Theoretical simulation and experimental research on the performance of air source EIICE under the heating conditions

The COP_t and PER_t are the main evaluation parameters of the performance of air source EIICE, they are defined as follows:

$$COP_t = \frac{Q_c + Q_g + Q_{cj}}{W_{cp}} \quad (20)$$

where the calculation method of C_d and A_{ev} is adopted from the literature [20].

2.3. The autonomous power supply system model

The model of power system can be described by the energy equilibrium [21] of the generator:

$$W_{gen,in} = W_{gen,out} + W_{gen,r} \quad (17)$$

The output power of generator is expressed by:

$$W_{gen,out} = 1.5(U_d I_d + U_q I_q + 2U_0 I_0) \quad (18)$$

The power losses of armature coil resistor are expressed by:

$$W_{gen,r} = 1.5(I_d^2 + I_q^2 + 2I_0^2)R \quad (19)$$

2.4. Numerical solution of the system model

After determining the structural parameters of each component, the relations of pipeline connecting as well as the condition parameters of air and refrigerant, the parameters of the whole system are converged to a certain range of accuracy by adjusting the condensing pressure, evaporating pressure and the throttle opening of internal combustion engine. Fig. 1 presents the flowchart of numerical algorithm of EIICE unit.

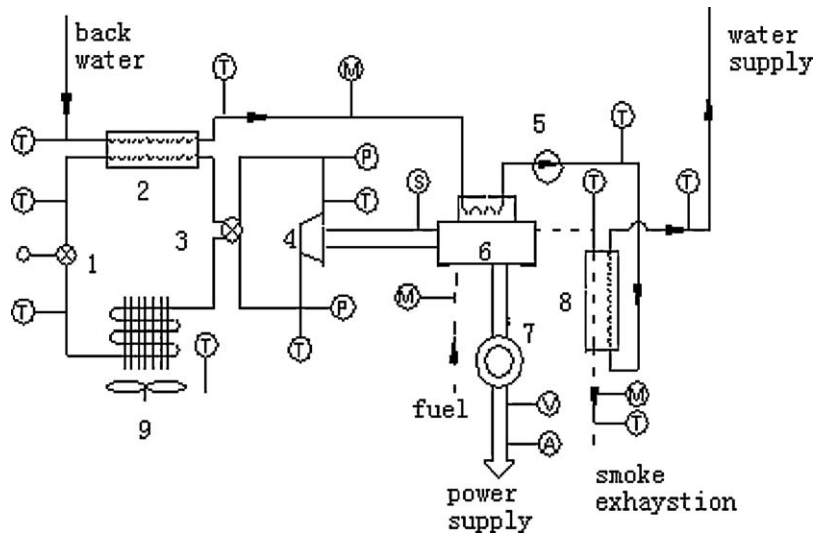


Fig. 2. Schematic diagram of EIICE system and measuring point. 1—EEV, 2—plate heat exchanger, 3—four-way valve, 4—compressor, 5—water pump, 6—internal-combustion engine, 7—generator, 8—exhaust gas heat exchanger, 9—outdoor heat exchanger, T—temperature measuring point, P—pressure measuring point, A—current measuring point, V—voltage measuring point, S—rotary speed measuring point, and M—flow rate measuring point.

$$PER_t = \frac{Q_c + W_{gen,out} + W_{pum} + Q_g + Q_{cj}}{Q_{fuel}} \quad (21)$$

4.1. The performance of air source EIICE under different rotary speeds

Figs. 3–6 present that the heat capacity of P-HE, heat recovered from CJ-HE, heat recovered from EG-HE, input power of compressor, output power of engine, fuel consumption, COP_t and PER_t vary with the rotary speed. Through the comparison of the simulated results and the experimental data, we can see that the variation trend of the simulated results is well-agreed with the experimental data under the same conditions. Fig. 3 shows that the heat capacity of P-HE is increased as the increase of the rotary speed. It is because that the power and the mass flow of refrigerant are increased with the increase of the rotary speed (Fig. 4). Fig. 3 also

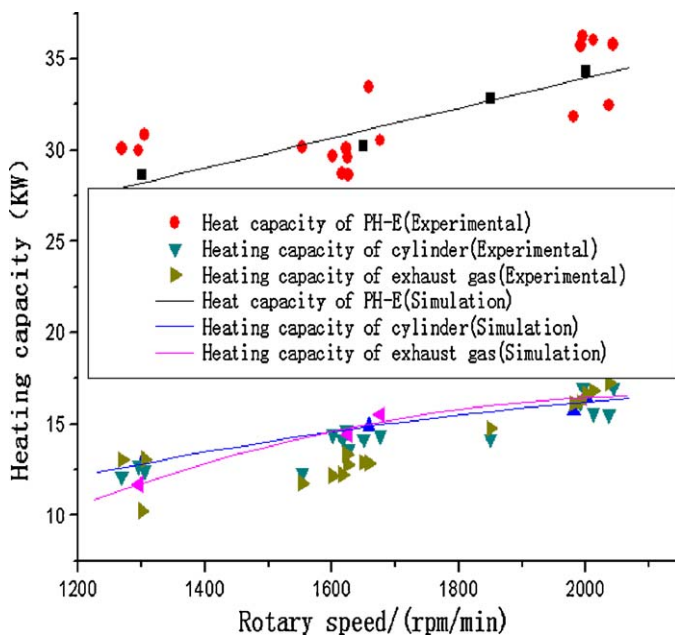


Fig. 3. Heat capacity of P-HE and heat recovered from cylinder jacket and exhaust gas variation with rotary speed.

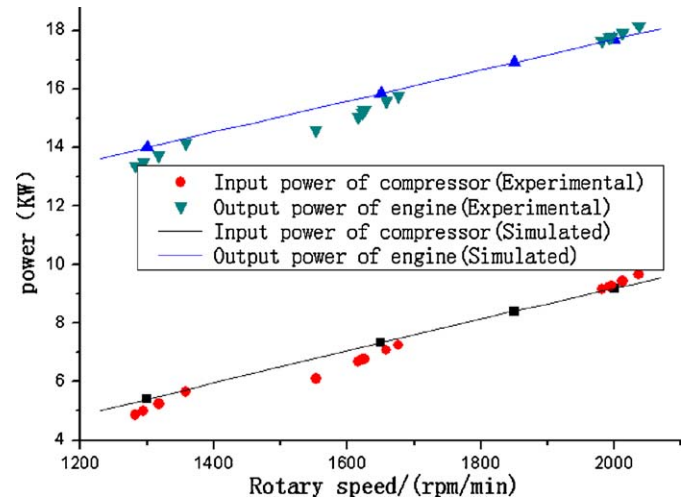


Fig. 4. The power of compressor and engine variation with rotary speed.

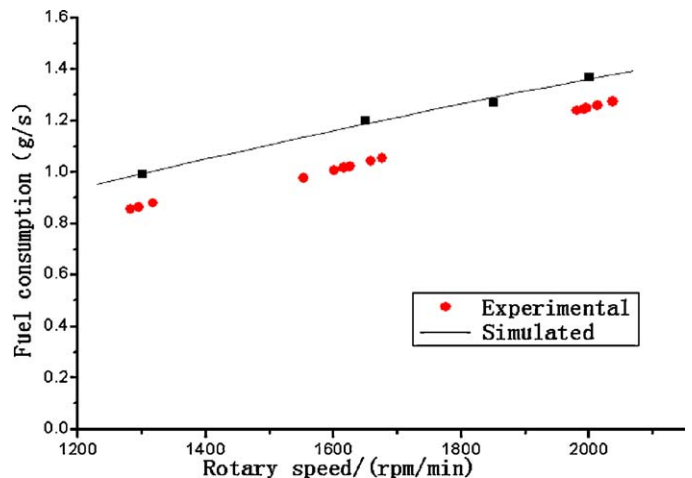


Fig. 5. Fuel consumption variation with rotary speed.

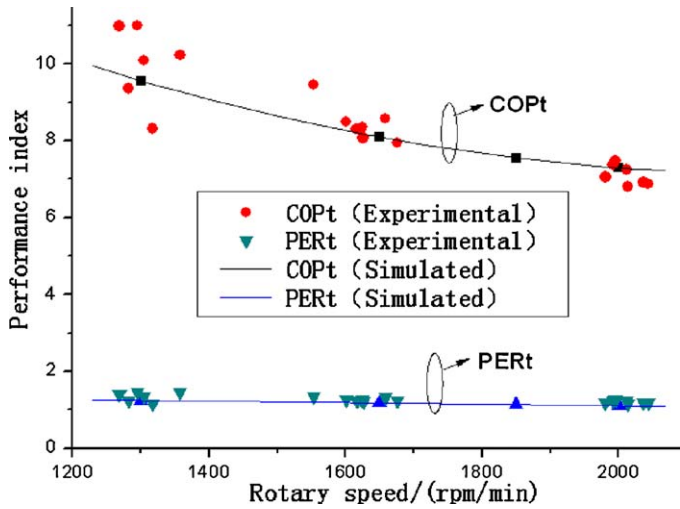


Fig. 6. COP_t and PER_t variation with rotary speed.

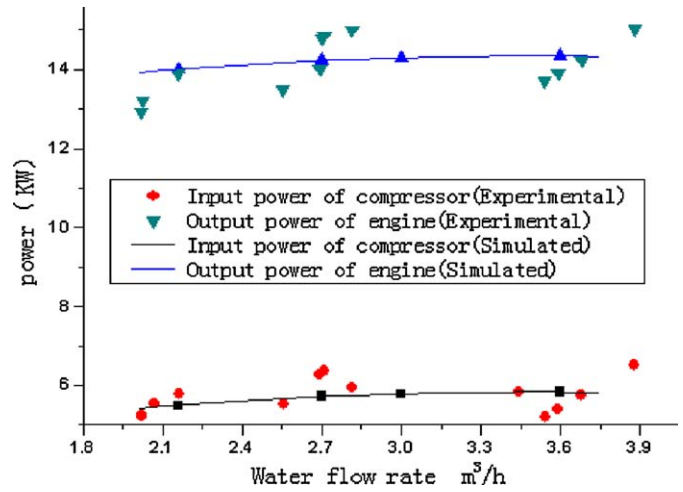


Fig. 8. The power of compressor and engine variation with water flow rate.

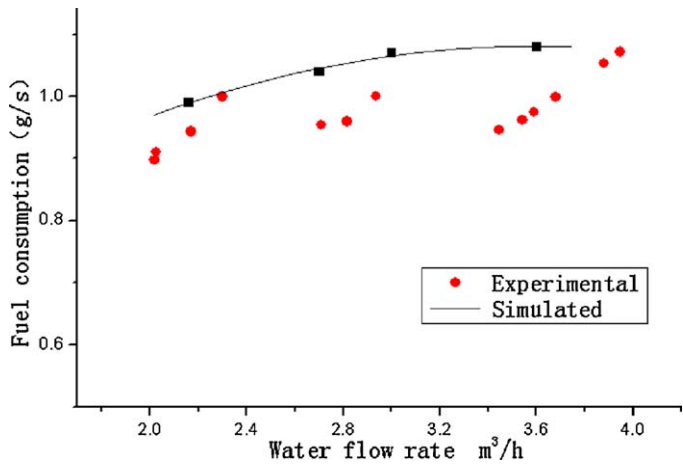


Fig. 9. Fuel consumption variation with water flow rate.

presents that heat recovered from CJ-HE and EG-HE is increased with the increase of the rotary speed, the reason is that the fuel consumption is increased with the increase of the rotary speed (Fig. 5), and the working substance in the cylinder and the temperature of the exhaust gas are also increased [22]. From Fig. 3, we also can see that the total waste heat recovery accounts for 90% of the P-HE's heat capacity and the waste heat recovery system has contributed to the improvement of EIICE's performance greatly. Therefore, this experimental apparatus has high heat recovery efficiency. This is also the advantage of the internal combustion heat pump relative to the electric driven heat pump. It is noted from Fig. 6 that, the COP_t and PER_t are decreased with the increase of the rotary speed. In addition, from the figures, we can find that the PER_t of the EIICE system varies little with the rotary speed, in other words, when the heating or cooling load changed acutely, the system can also be operated in a stable efficiency condition. That is to say, the EIICE system has a good speed control performance.

4.2. The performance of air source EIICE under different water flow rates of P-HE

Figs. 7–10 present that the heat capacity of P-HE, heat recovered from CJ-HE, heat recovered from EG-HE, input power of compressor,

output power of engine, fuel consumption, COP_t and PER_t vary with the water flow rate. From these figures, we can see that the variation trend of the simulated results is consistent with experimental data under the same conditions. From Fig. 7, we can see that under the condition of a fixed rotary speed, the bigger the

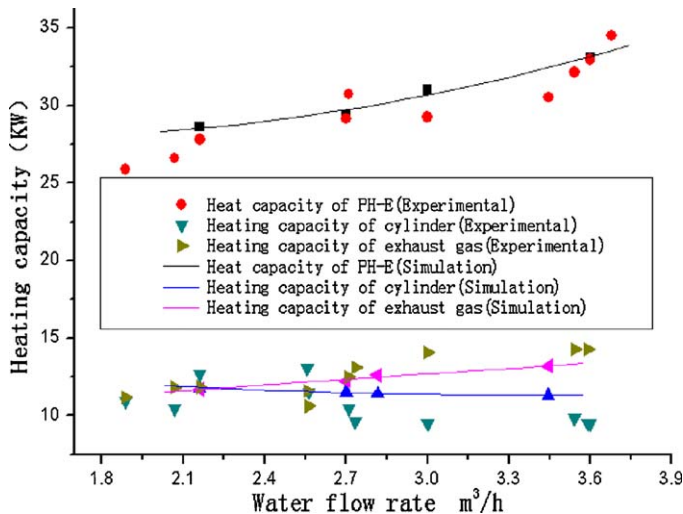


Fig. 7. Heat capacity of P-HE and heat recovered from cylinder jacket and exhaust gas variation with water flow rate.

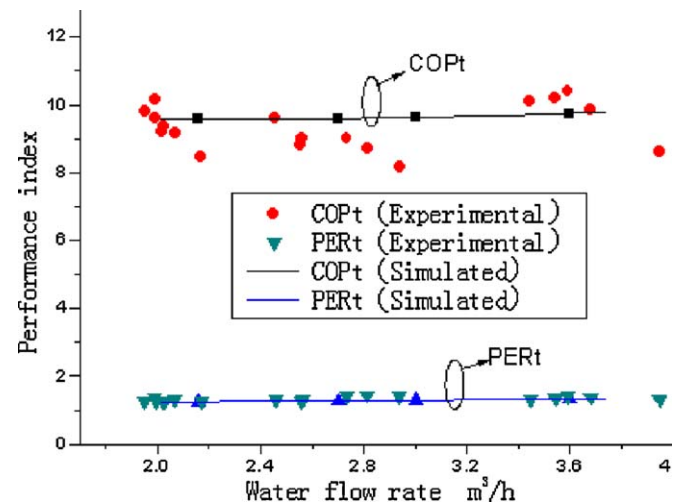


Fig. 10. COP_t and PER_t variation with water flow rate.

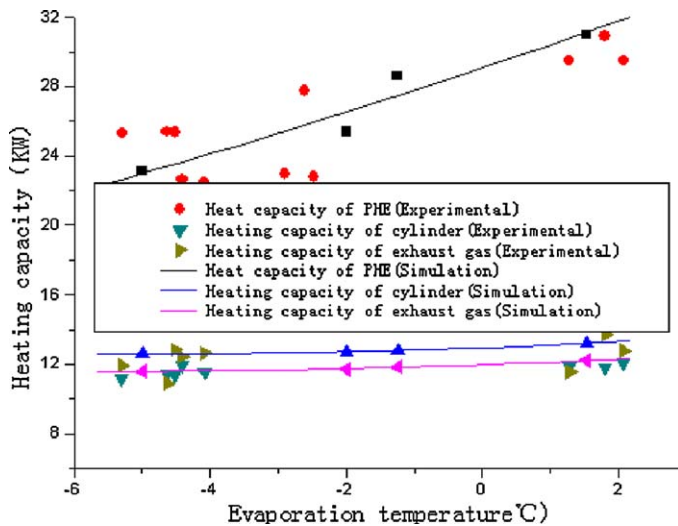


Fig. 11. Heat capacity of P-HE and heat recovered from cylinder jacket and exhaust gas variation with evaporation temperature.

water flow rate is, the greater the heat capacity of the P-HE is, it is because that the heat transfer coefficient of the P-HE is increased with the increase of the water flow rate. Increasing the water flow rate could decrease the heat transfer temperature difference and the losses of the P-HE. It is noted from Fig. 7 that with the increase of the water flow rate, the heat recovered from CJ-HE decreases in a small range. The reason is mainly about that the heat recovered from the cylinder jacket is bounded by the thermostat. The increase of the water flow rate of the P-HE leads to the increase of its heat transfer efficiency, which will make the inlet water temperature of the expansion tank lower, and then the water temperature of whole expansion tank will get lower, thus, the inlet water temperature of the cylinder could be decreased. The thermostat adjusts the degree of its openness to reduce the water flow rate to ensure that the temperature of the cylinder is nearly unchanged. Therefore, the heat recovered from CJ-HE will decrease with the increase of the water flow rate. When the terminal load is increased with the increase of the water flow rate of P-HE, the power of the compressor and the fuel consumption of the engine will increase (Figs. 8 and 9), leading to the increased temperature of the exhaust gas, therefore the heat recovered from EG-HE is gradually increased with the increase of the water flow rate of P-HE which can see from Fig. 7. As shown in Fig. 10, it is obvious that with the increase of the water flow rate, COP_t and PER_t are all increased at a small range. The reason is that the efficiency of P-HE increased as the increase of its water flow rate, as a result, the performance of the whole system is improved.

4.3. The performance of air source EIICE under different evaporation temperatures

Figs. 11–14 present that the heat capacity of P-HE, heat recovered from CJ-HE, heat recovered from EG-HE, input power of compressor, output power of engine, fuel consumption, COP_t and PER_t vary with the evaporation temperature. From the figures, we can confirm that the variation trend of the theoretical simulation is compatible with experimental result under the same conditions. From Fig. 11, we can see that the heat capacity of P-HE increased with the increase of the evaporation temperature, it is because that under the conditions of a fixed condenser temperature, the higher evaporation temperature is, the greater power consumption and refrigerant flow rate are. As shown in Fig. 11, the heat recovered from CJ-HE and EG-HE will increase if the evaporation temperature is increased, the main reason is that the power consumption

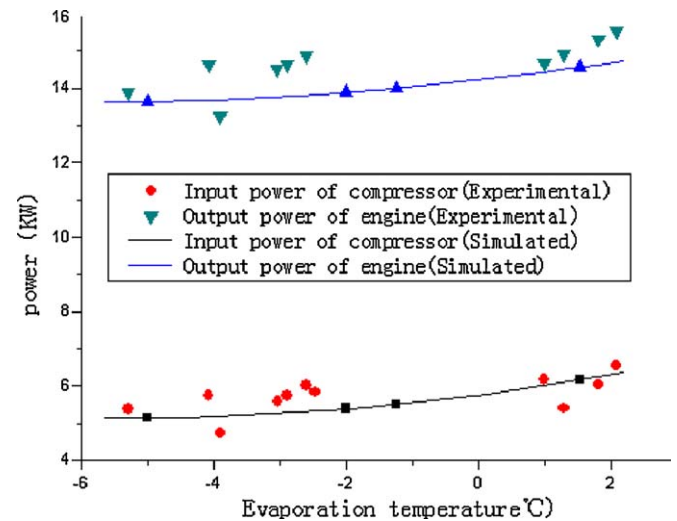


Fig. 12. The power of compressor and engine variation with evaporation temperature.

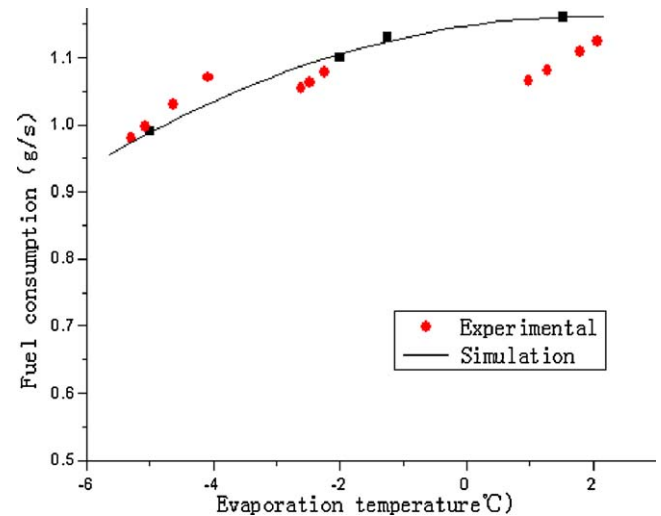


Fig. 13. Fuel consumption variation with evaporation temperature.

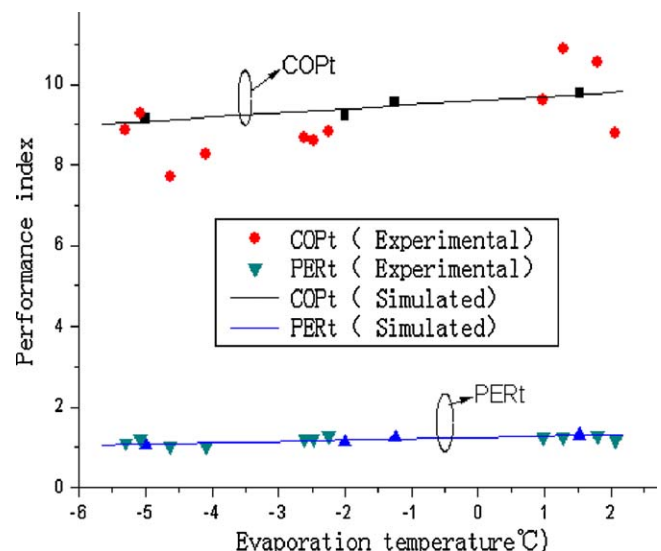


Fig. 14. COP_t and PER_t variation with evaporation temperature.

of the compressor and engine increase with the increase of the evaporation temperature (Fig. 12), and the fuel consumption is also increased (Fig. 13), therefore the working medium in the cylinder and exhaust gas temperature are also increased [22], which results in the heat recovered from CJ-HE and EG-HE increase. It is found from Fig. 14 that the COP_t and PER_t are also increased with the increase of the evaporation temperature.

5. Conclusions

- (1) Through the comparison of the simulated results and the experimental data, we can see that the variation trend of the simulated results is well-agreed with the experimental data under the same conditions. And the relative deviation between the theoretical results and experimental data is mostly less than 10%. It can be concluded that the model is capable of simulating the performance of air source EIICE system with the acceptable accuracy.
- (2) The experimental and numerical results indicate that the heat capacity of P-HE, heat recovered from EG-HE, input power of compressor, output power of engine and fuel consumption are increased with the increase of the rotary speed, water flow rate of P-HE and evaporation temperature. Heat recovered from CJ-HE would increase as the rotary speed and evaporation temperature increases, respectively, but decrease as the water flow rate of P-HE increases. The COP_t and PER_t of air source EIICE system are also increased with the increase of the water flow rate of P-HE and the evaporation temperature, but decreased with the increase of the rotary speed.

Acknowledgements

Supported by the Hi-tech Research and Development Program of China (2007AA05Z223), and by NSFC, National Education Department for Doctor Center Foundation (200800560041), National Natural Science Foundation of China (Grant No. 51076112).

References

- [1] J.L.W. Vinton, R.P. Getman, Gas engine heat pump performance in a southern climate, *ASHRAE Transactions* 101 (1995) 1389–1395.
- [2] Z.J. Xu, Z. Yang, Saving energy in heat-pump air conditioning system driven by gas engine, *Energy and Buildings* 41 (2009) 206–211.
- [3] S. Sanaye, M.A. Meybodi, M. Chahartaghi, Modeling and economic analysis of gas engine heat pumps for residential and commercial buildings in various climate regions of Iran, *Energy and Buildings* 42 (2010) 1129–1138.
- [4] R.R. Zhang, X.S. Lu, S.Z. Li, W.S. Lin, A.Z. Gu, Analysis on the heating performance of a gas engine driven air to water heat pump based on a steady-state model, *Energy Conversion and Management* 46 (2005) 1714–1730.
- [5] Z. Yang, S.G. Zhang, Dynamic study of the exhaust heat recovery system for a gas engine-driven heat pump, *Acta Energiae Solaris Sinica* 25 (2004) 712–716.
- [6] S.Z. Li, W.G. Zhang, R.R. Zhang, D.X. Lv, Z. Huang, Cascade fuzzy control for gas engine driven heat pump, *Energy Conversion and Management* 46 (2005) 1757–1766.
- [7] Z. Yang, H.B. Zhao, Z. Fang, Modeling and dynamic control simulation of unitary gas engine heat pump, *Energy Conversion and Management* 48 (2007) 3146–3153.
- [8] Y.L. Li, X.S. Zhang, L. Cai, A novel parallel-type hybrid-power gas engine-driven heat pump system, *International Journal of Refrigeration* 30 (2007) 1134–1142.
- [9] S. Yagyu, I. Fujishima, Y. Fukuyama, T. Morikawa, N. Obata, Performance characteristics of a gas engine driven Stirling heat pump, in: *Energy Conversion Engineering Conference and Exhibit (IECEC) 35th Intersociety*, 2000, pp. 85–91.
- [10] S. Yagyu, I. Fujishima, J. Corey, N. Isshiki, I. Satoh, Design, simulation and test results of a heat-assisted three-cylinder Stirling heat pump (C-3), in: *Energy Conversion Engineering Conference (IECEC) Proceedings of the 32nd Intersociety*, 1997, pp. 1033–1038.
- [11] J.H. Yang, J.K. Dou, *The Technology of Internal Combustion Engine Performance*, People's Communications Press, Beijing, 2000.
- [12] Y.Y. Yan, H.C. Li, T.F. Lin, Condensation heat transfer and pressure drop of refrigerant R134a in a plate heat exchanger, *International Journal of Heat and Mass Transfer* 42 (1999) 993–1006.
- [13] A. Muley, R.M. Manlik, Experimental study of turbulent flow heat transfer and pressure drop in a plate heat exchanger with chevron plates, *Journal of Heat Transfer* 121 (1999) 110–117.
- [14] Y.Y. Yan, T.F. Lin, B.C. Yang, Evaporation heat transfer and pressure drop of refrigerant R134a in a plate heat exchanger, *American Society of Mechanical Engineers (Paper)* (1999).
- [15] H. Huang, Study of the dynamic performance and frosting, defrosting cycle of air cooled heat pump/chiller, Doctoral Dissertation, Xi'an, Xi'an University, 1999.
- [16] P.Y. Wang, X.G. Yuan, Computer simulation of car air-conditioning cooling system, *Journal of Beijing University of Aeronautics and Astronautics* 233 (1997) 590–595.
- [17] T. Nitheanandan, H.M. Soliman, R.E. Chan, A proposed approach for correlating heat transfer during condensation inside tubes, *ASHRAE Transactions* 96 (1990) 230–241.
- [18] G.L. Ding, C.L. Zhang, *Simulation and Optimization of Refrigeration and Air-conditioning Device*, Scientific Publishing House, Beijing, 2001.
- [19] S.X. Zhang, Numerical simulation on evaporator test of cooling system, Master Thesis, Nanjing, Nanjing University of Aeronautics and Astronautics, 2001.
- [20] W.B. Wen, J.Z. Wang, Experimental research on electronic expansion valve refrigerant flow characteristics, *Fluid Machinery* 26 (1998) 25–30.
- [21] Y. Tang, Y.H. Zhang, Y. Fan, *Dynamic Analysis of AC Motor*, Mechanical Industry Press, Beijing, 2005.
- [22] H.B. Zhao, Theoretical and experimental research on system of air source autonomous power supply gas engine-driven heat pump, Doctoral Dissertation, Tianjin, Tianjin University, 2007.